A Methodology for Protective Vibration Monitoring of Hydropower Units Based on the Mechanical Properties

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It is important to monitor the radial loads in hydropower units in order to protect the machine from harmful radial loads. Existing recommendations in the standards regarding the radial movements of the shaft and bearing housing in hydropower units, ISO-7919-5 (International Organization for Standardization, 2005, "ISO 7919-5: Mechanical Vibration—Evaluation of Machine Vibration by Measurements on Rotating Shafts—Part 5: Machine Sets in Hydraulic Power Generating and Pumping Plants," Geneva, Switzerland) and ISO-10816-5 (International Organization for Standardization, 2000, "ISO 10816-5: Mechanical Vibration—Evaluation of Machine Vibration by Measurements on Non-Rotating Parts—Part 5: Machine Sets in Hydraulic Power Generating and Pumping Plants," Geneva, Switzerland), have alarm levels based on statistical data and do not consider the mechanical properties of the machine. The synchronous speed of the unit determines the maximum recommended shaft displacement and housing acceleration, according to these standards. This paper presents a methodology for the alarm and trip levels based on the design criteria of the hydropower unit and the measured radial loads in the machine during operation. When a hydropower unit is designed, one of its design criteria is to withstand certain loads spectra without the occurrence of fatigue in the mechanical components. These calculated limits for fatigue are used to set limits for the maximum radial loads allowed in the machine before it shuts down in order to protect itself from damage due to high radial loads. Radial loads in hydropower units are caused by unbalance, shape deviations, dynamic flow properties in the turbine, etc. Standards exist for balancing and manufacturers (and power plant owners) have recommendations for maximum allowed shape deviations in generators. These standards and recommendations determine which loads, at a maximum, should be allowed before an alarm is sent that the machine needs maintenance. The radial bearing load can be determined using load cells, bearing properties multiplied by shaft displacement, or bearing bracket stiffness multiplied by housing compression or movement. Different load measurement methods should be used depending on the design of the machine and accuracy demands in the load measurement. The methodology presented in the paper is applied to a 40 MW hydropower unit; suggestions are presented for the alarm and trip levels for the machine based on the mechanical properties and radial loads. [DOI: 10.1115/1.4023668]

Keywords: vibration, bearing, load, hydropower

1 Introduction

Different methods are used to monitor and protect the hydropower unit from harmful operation modes. Eccentricities and shape deviations in generators [1], mass unbalances in rotors [2], dynamic flow properties in turbines [3], etc. cause radial loads on the bearings and supporting structures. Figure 1 presents the components in a hydropower unit. Demands on the unit's instrumentation are dependent upon the importance of the unit and the cost of outage. Some older units do not have an installed vibration monitoring system, but most units are equipped with shaft displacement sensors or accelerometers to monitor vibrations in the unit. There are also units equipped with air gap sensors that measure the distance between the generator's rotor and stator. In some hydropower units, load cells are installed inside the bearing or bearing bracket in order to monitor radial loads on the structure. Several different types of sensors and methods can be used to monitor vibrations in a hydropower unit; the key issue with vibration monitoring is to protect the machine and avoid outage.

The recommendations in various standards for permitted vibration level values are often used as an aid to determine and decide if a unit is to be stopped for maintenance. The standards ISO 7919-5 [4] and ISO 10816-5 [5] divide vibration levels into classes with increasing levels from Class A to Class D, where Class A is a good machine that does not need attention while Class D is a machine that should be stopped for immediate corrective action. The permitted levels for each class vary with the unit's rotational speed; a low speed permits higher values of vibration levels in each class, compared to high speed. According to Totir et al. [6], ISO 7919-5 and ISO 10816-5 are not sufficient as vibration monitoring standards. The recommended vibration levels for each class are based on the unit's rotational speed and statistical data; consideration of the physical properties of bearings and brackets is not taken, which strongly affect the relationship between the radial load and vibration levels and which vibration levels the components can withstand.

The objective of this paper is to present a vibration monitoring methodology based on the physical properties of the hydropower

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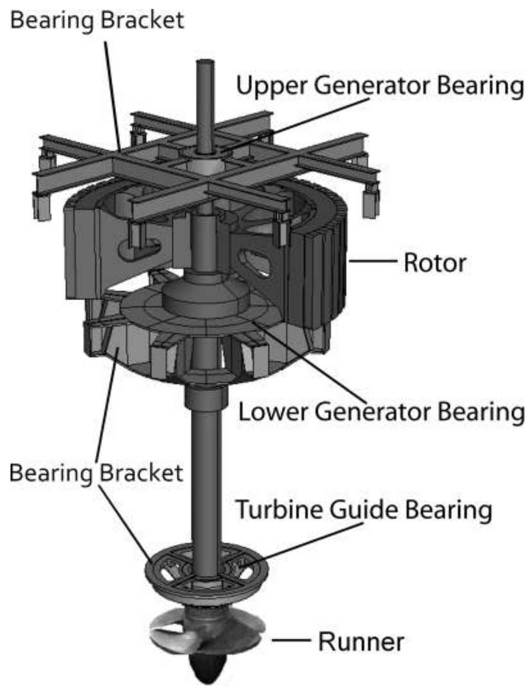

Fig. 1 Components in a hydropower unit

units, i.e., based on the design criteria of the hydropower unit and the measured radial loads in the machine during operation.

The structure of this paper is to first describe the theoretical background regarding how to determine vibrations and forces in hydropower units. Following this, an example of design criteria for hydropower units is presented. Subsequently, Secs. 4 and 5 present a proposed methodology for vibration monitoring and an example of how the methodology is applied to a hydropower unit. Finally, discussions and conclusions regarding the methodology are presented.

2 Radial Forces and Vibrations in Hydropower Units

In order to evaluate the radial movement's influence on the structure, the stiffness (and in some cases, damping) properties should be considered. In a vertical hydropower unit, the radial forces that influence the radial bearing and bracket are ideally equal to zero. The pressure generation that acts on the bearing segments is balanced internally in the bearing housing and does not influence the surrounding structure. Unfortunately, there are no ideal machines. All machines are more or less affected to some degree by deviations in geometry, balancing of the rotor, stator and rotor eccentricity, dynamic flow properties in the turbine, etc.

2.1 Interaction Between the Bearing and the Bearing Bracket. In situations where vibration measurement is performed with accelerometers mounted on bearing housings connected to stiff brackets and the stiffness properties of the bearing and bracket are neglected, the vibration level of the shaft will be greatly underestimated. In situations where the measurement is performed with displacement sensors measuring the distance between shaft and housing in a machine with a stiff bearing, an underestimation of the shaft movement will occur, since the greatest displacement will be in the bracket. It is important to consider the combined stiffness and damping properties, not only the bearing or bracket properties. Figure 2 presents a schematic description of the bearing and bracket where k_{ij} is the stiffness parameter of the bearing, c_{ij} is the damping parameter of the bearing, and h_{ij} is the stiffness parameter of the bracket. The displacement vector $\mathbf{u}$ rotates with the frequency Ω in the fixed coordinate system oriented at the geometrical center of the bearing.

The combined stiffness and damping properties (fluid film properties in the bearing combined with the bearing bracket properties)

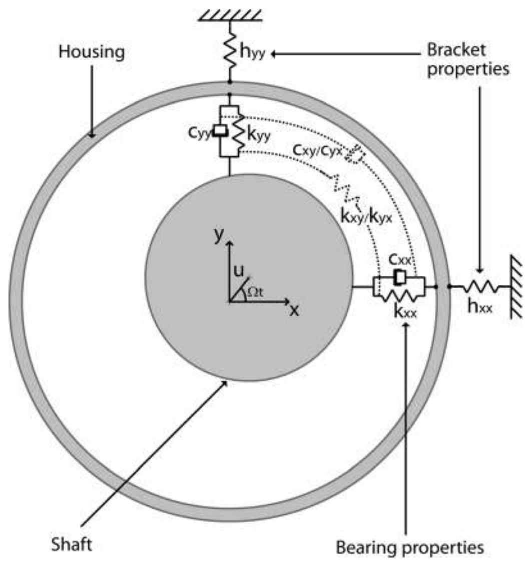

Fig. 2 Schematic figure presenting the bearing and brackets in a hydropower unit

are calculated using the impedance method [7]. In Eq. (2), assume that a force acting on the bearing, caused by the rotor, is represented as f and bearing stiffness as k and bearing damping as c. Absolute displacement of shaft and housing is presented as $\mathbf{u}_S$ and $\mathbf{u}_H$; the shaft displacement in the bearing (i.e., shaft displacement relative to bearing center) is $\mathbf{u} = \mathbf{u}_S - \mathbf{u}_H$. The relationship between bearing load and bearing properties will be treated later in the paper.

The force equilibrium model for the bearings oil film properties is formulated as

$$\begin{Bmatrix} f_x \\ f_y \end{Bmatrix} = \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix} \begin{Bmatrix} u_x \\ u_y \end{Bmatrix} + \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix} \begin{Bmatrix} \dot{u}_x \\ \dot{u}_y \end{Bmatrix} + \begin{bmatrix} m_{xx} & m_{xy} \\ m_{yx} & m_{yy} \end{bmatrix} \begin{Bmatrix} \ddot{u}_x \\ \ddot{u}_y \end{Bmatrix} \tag{1}$$

In most conventional bearing models, the fluid inertia forces are disregarded [8]. The bearings' fluid inertia properties will be disregarded in the remainder of this paper. For a purely harmonic motion with a whirl frequency of Ω the force equilibrium model for the bearings oil film properties is formulated as [9]

$$\begin{Bmatrix} f_x \\ f_y \end{Bmatrix} = \begin{bmatrix} k_{xx} + i\Omega c_{xx} & k_{xy} + i\Omega c_{xy} \\ k_{yx} + i\Omega c_{yx} & k_{yy} + i\Omega c_{yy} \end{bmatrix} \begin{Bmatrix} u_{Sx} - u_{Hx} \\ u_{Sy} - u_{Hx} \end{Bmatrix} = \mathbf{Z}(\mathbf{u}_S - \mathbf{u}_H) \tag{2}$$

Radial loads $\mathbf{f}$ acting on the bearing also act on the bearing bracket; the relationship between force and displacement, for a bearing bracket with stiffness $\mathbf{H}$, is presented in Eq. (3)

$$\begin{Bmatrix} f_x \\ f_y \end{Bmatrix} = \begin{bmatrix} h_{xx} & h_{xy} \\ h_{yx} & h_{yy} \end{bmatrix} \begin{Bmatrix} u_{Hx} \\ u_{Hy} \end{Bmatrix} \tag{3}$$

The damping and the cross-coupled stiffness in the bearing bracket are neglected; the bearing brackets are made of steel beams. The relationship between the displacement and combined stiffness and damping properties can be written in short form as

$$\left.\begin{aligned} &\mathbf{f} = \mathbf{Z}(\mathbf{u}_S - \mathbf{u}_H) = \mathbf{Z}\mathbf{u}_S - \mathbf{Z}\mathbf{u}_H \\ &\mathbf{f} = \mathbf{H}\mathbf{u}_H \ \Rightarrow \mathbf{u}_H = \mathbf{H}^{-1}\mathbf{f} \end{aligned}\right\}$$
$$\mathbf{f} = \mathbf{Z}\mathbf{u}_S - \mathbf{Z}\mathbf{H}^{-1}\mathbf{f} \Rightarrow \left[\mathbf{I} + \mathbf{Z}\mathbf{H}^{-1}\right]\mathbf{f} = \mathbf{Z}\mathbf{u}_S \Rightarrow \mathbf{f} = \left[\mathbf{I} + \mathbf{Z}\mathbf{H}^{-1}\right]^{-1}$$
$$\mathbf{Z}\mathbf{u}_S = \mathbf{D}\mathbf{u}_S$$
$$\mathbf{D} = \left[\mathbf{I} + \mathbf{Z}\mathbf{H}^{-1}\right]^{-1}\mathbf{Z} \tag{4}$$

where $\mathbf{D} = \mathbf{K}_c + i\Omega\mathbf{C}_c$ is the combined properties of the bearing and bracket, $\mathbf{K}_c$ is the combined stiffness, and $\mathbf{C}_c$ is the combined damping.

If the stiffness is assumed to be linear in both the bearing and the bracket, the shaft's displacement will be distributed between the bearing and the bracket, according to Fig. 3. Figure 3 also shows the change in the combined damping, in relation to the bearing's damping, depending of the stiffness relationship between the shaft and bracket. The abscissa shows the stiffness relationship between the bearing and the bracket. In a hydropower unit with a design according to Fig. 1, the stiffness of the bearing bracket varies between 0.2 GN/m to 4 GN/m, depending on how the inner and outer parts of the bracket are connected and the dimensions of the steel structure. The bearing properties at "normal" operation for machines, according to Fig. 1, are k_{xx}, $k_{yy} \sim$ 0.4-2 GN/m and the cross coupled stiffness terms k_{xy} and k_{yx} are significantly less than k_{xx} and k_{yy} for tilting pad bearings [10]. The damping c_{xx} and c_{yy} in the bearing is set to 0.1 GN s/m in the example presented in Fig. 3. These assumed values of the bearing properties are strongly dependent on the bearing clearance and operation conditions (i.e., bearing load). The values can deviate far from these assumed ones, but in order to visualize the importance of the bearing and bracket properties, these values were used to generate Fig. 3.

The functions in Fig. 3 shows the distribution of the shaft's total displacement between the shaft's movement in the bearing and the bearing housing movement, i.e., u_{sj}/u_j, respectively, u_{hj}/u_j. Figure 3 also presents the relationship between total damping and the bearing's damping, i.e., c_{cjj}/c_{jj}. The aforementioned reasoning is very simplified and does not take into consideration the fact that a journal bearing is nonlinear and that the stiffness change greatly depends upon for which eccentricity the stiffness was calculated. This demonstrates the difficulty of evaluating vibration data collected from displacement sensors or accelerometers without knowing the properties of the bearing and bracket.

2.2 Radial Forces in a Hydropower Unit. The rotor in a hydropower unit will influence the radial bearing with forces that can be both static and dynamic. Static loading of the bearing is not possible to detect with accelerometers since the static forces do not create vibrations in the structure.

Static forces caused by, e.g., large stator eccentricity can, however, result in large forces on the bearing, supporting structure, and stator. The forces vary, though, at the unit's starts and stops and thereby contribute to the fatigue of the components in the unit. Dynamic loading of the structure is easy to detect with modern vibration equipment since a varying force gives rise to displacements that can be detected by using accelerometers or displacement sensors.

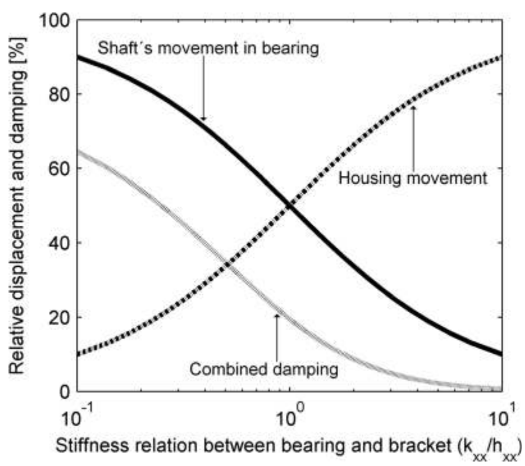

Fig. 3 Total damping and distribution of motions in the bearing and the bracket

Different methods have been used to determine the radial forces in vertical hydropower units. For "direct" force determination, load cells have been installed behind pivot pins [11], strain gauges have been installed inside bearing pivot pins [12], and strain gauges have been installed on the bearing brackets [13]. Forces can also be determined from bearing housing movements (measured with accelerometers) multiplied by the stiffness of the bracket connecting the housing to the concrete structure. Radial forces determined from accelerometer measurements only include dynamic loads. It is not possible to measure the static radial load by using accelerometers.

2.3 Determination of Bearing Loads Using Shaft Displacement Measurements. A method using the bearing properties and shaft movements has also been developed in order to determine the radial loads in the guide bearing in hydropower units. Radial guide bearings in large hydropower units are hydrodynamic journal bearings, often of the tilting pad type.

As presented earlier in this paper, the force in the bearing is caused by relative movements between the shaft and bearing housing. The dynamic properties of a bearing, i.e., $\mathbf{K}$ and $\mathbf{C}$, are dependent on the bearing geometry, properties of the lubricant, rotational speed, eccentricity, etc. The shaft eccentricity ε in a radial bearing is the relationship between the radial shaft displacement and the radial bearing clearance. At small eccentricities ($\varepsilon < 0.6$), the bearing properties of journal bearings used in hydropower units can be considered as linear and there are no analytical expressions to determine the dynamic properties for tilting pad bearings [9,10]. To determine the dynamic properties of tilting pad bearings, numerical calculations are required. Bearing dynamics software often enables bearing parameters to be calculated at a prescribed bearing load or journal eccentricity. Figures 4(*a*) and 4(*b*) present the results from a calculation of the direct stiffness and damping properties (k_{xx} and c_{xx}) as a function of eccentricity at a fixed bearing clearance and rotational speed for a tilting pad bearing in a hydropower unit. The calculations were performed in RAPPID-RDA™, which is a commercial rotordynamic analysis software.

If all of the bearing properties are calculated, i.e., also k_{xy}, k_{yx}, k_{yy}, $c_{xy} \ldots c_{yy}$, $m_{xx} \ldots m_{yy}$, the calculated bearing load as a function of eccentricity will be as presented in Fig. 5, according to Eq. (1) (the load is applied in the x-direction).

To determine the bearing load from a measured shaft displacement and calculated bearing parameters, the bearing parameters for the present bearing clearance and the shaft's displacement relative to the bearing center must be known. The bearing clearance changes depending on the temperature of the bearing, surrounding structures, and shaft. Using four displacement sensors at each bearing, installed with 90 deg separation, it is possible to compensate for thermal changes and determine the center of the bearing. The bearing center and clearance are determined by using a hydraulic jack to push the shaft in the $+x$, $-x$, $+y$, and $-y$ directions and using the sensors to register the center position and bearing clearance. Figure 6 presents the positions of the sensors and the thermal changes that influence the bearing clearance.

When the bearing clearance is measured on a hydropower unit and the symmetric shape variation due to temperature and external forces are assumed, then both of the shaft displacements from the center and the change in the bearing clearance can be described. In Fig. 6 c_{mx} represents the measured radial bearing clearance in the x-direction. When the temperature in the bearing changes the bearing clearance also changes, due to the changed shaft and bearing diameter. From Fig. 6, the present radial bearing clearance c_{bx} can be determined by adding and subtracting the geometrical changes of the bearing Δd_b and the shaft Δd_s to the measured bearing clearance, i.e., $c_{bx} = c_{mx} + \Delta d_b - \Delta d_s$. When the shaft is displaced the distance x in the x-direction (see Fig. 6), the distance between the bearing surface and the shaft surface at sensor 1 is $c_{bx} - \mathrm{x}$ and $c_{bx} + \mathrm{x}$ at sensor 3. Half of the sum of sensor 3 and

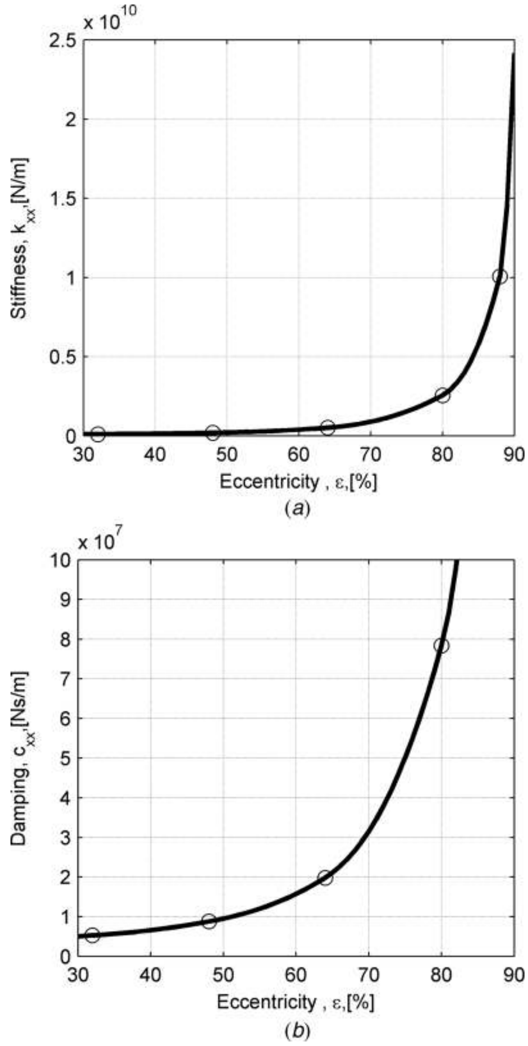

Fig. 4 (*a*), (*b*) Stiffness and damping properties as a function of eccentricity for a tilting pad bearing in the hydropower unit

sensor 1 then represent the radial bearing clearance $(c_{bx}+x+c_{bx}-\mathrm{x})/2=c_{bx}$. Half of the difference between sensor 3 and sensor 1 provides the shaft's displacement from the center of the bearing $((c_{b\ x}+x)-(c_{bx}-\mathrm{x}))/2=x$. Using the corresponding reasoning, the displacement of the y-axis can also be obtained.

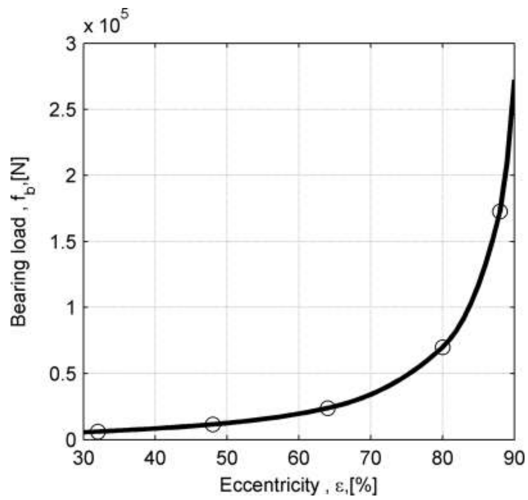

Fig. 5 Bearing load as a function of eccentricity

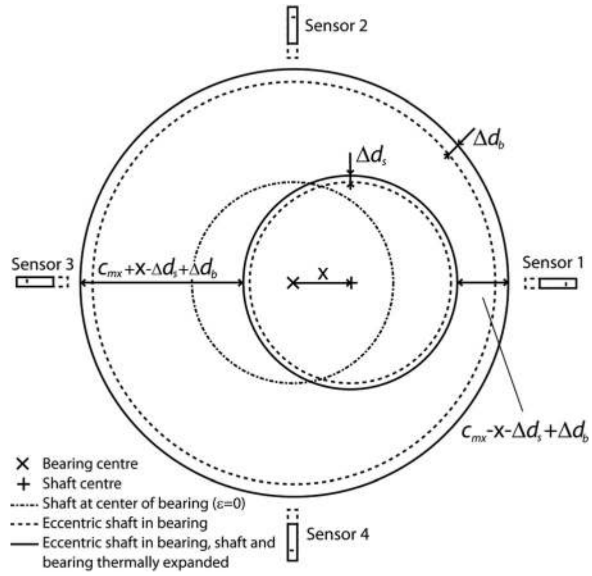

Fig. 6 Position of the displacement sensors and the thermal change's influence on the bearing clearanc

The changes in the bearing clearance due to thermal changes are normally less than 15% of the bearing clearance, but the changes still influence the bearing parameters and must be taken into consideration. Table 1 presents an example of how the bearing clearance changes due to the thermal properties of the machine. The time 2:00 p.m. in Table 1 corresponds to the machine start-up after being out of operation overnight (cold machine); at the other points of time in Table 1 the machine has been in operation for several hours.

By knowing the radial shaft displacement in the bearing, the present bearing clearance, and the bearing properties at these eccentricities, the load can be determined, provided that the magnitude of the eccentricity is decisive for the bearing properties, not the relationship between the static and dynamic parts of the eccentricity. By calculating the eccentricity of the shaft in the x- and y-directions the total eccentricity of the shaft ε_t and the phase θ will be as calculated in Eq. (5)

$$\begin{aligned}\varepsilon_t &= \sqrt{\left(\frac{2x}{c_{bx}}\right)^2+\left(\frac{2y}{c_{by}}\right)^2}\\ \theta &= \arctan\left(\frac{y}{x}\right)+n\pi \quad (n=0 \text{ if } x\geq 0, n=1 \text{ if } x<0)\end{aligned} \tag{5}$$

Figure 7 presents the bearing load as a function of the eccentricity and bearing clearance. By generating a function or look-up table for the data in Fig. 7, the load is determined by inserting the bearing clearance and eccentricity.

2.4 Measured Bearing Load and Bearing Load Calculated From Shaft Displacement and Bearing Properties.

Figure 8 presents a comparison between the measured and calculated bearing load at the upper generator bearing for a 40 MW hydropower unit at synchronous operation. The bearing load was measured

Table 1 Measured bearing clearance in the *x*-direction

Time	Bearing clearance (mm)
2.00 p.m.	0.41
7.00 p.m.	0.38
9.00 p.m.	0.39
10.30 p.m.	0.37

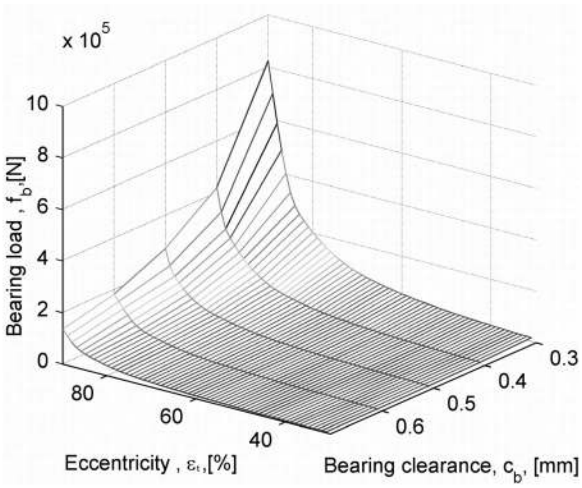

Fig. 7 Example of the calculated bearing load as a function of the bearing clearance and eccentricity for the tilting pad in a hydropower unit

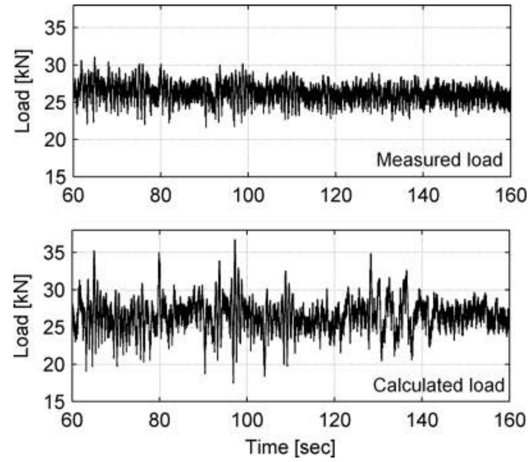

Fig. 8 Bearing load determined using strain gauges inside the pivot pin (upper figure) and the shaft displacement multiplied with the bearing properties (lower figure)

using strain gauges installed inside the pivot pin and is presented as the upper figure in Fig. 8. The shaft displacement was measured using displacement sensors and the load-eccentricity-clearance relation presented in Fig. 7 is calculated for the upper generator bearing of the 40 MW unit. The measurement was performed at 9:00 p.m. with a corresponding bearing clearance of 0.39 mm, as shown in Table 1. To determine the bearing load from the measured shaft displacement, the load-eccentricity-clearance relation in Fig. 7 is multiplied with the eccentricity of the shaft. The lower figure in Fig. 8 presents the bearing load calculated from the shaft displacement and bearing properties.

3 Design Criteria for Mechanical Components in a Hydropower Unit

When new machines are manufactured, the components are designed according to specific design criteria. Several of the mechanical components in a hydro-electric power plant are constructed for loads that vary according to factors such as power, temperature, starts and stops, hydraulic loads, unbalances, its own weight, and faults that may occur.

3.1 Fatigue. The design criteria of the unit stipulate how the unit is to be used and how many times it can be started and stopped during its technical lifetime. The number of cycles and events provides input data for the load spectra that is used as support for the fatigue design of the unit. Examples of load components included in the spectra are presented in Table 2.

Using the load spectra determined by the customer, the machine is designed considering its structural strength and fatigue. In order to avoid damage to the mechanical components, it is important to know the design load for the components and to monitor the loads that occur in critical components. Regarding the damage caused by high loads from the rotating structure, the damage often occurs in the bearings, welding in the bearing brackets, shaft couplings, and the interconnection. When a new machine is designed, the manufacturer performs a finite element analysis (FEA) and fatigue analysis based on the customer's load spectra. In order to protect the machine from harmful loads, it is important that the customer retain information regarding these loads. For old machines where fatigue calculations are not available, the critical components should be identified using the FEA and the critical loads determined using fatigue analysis. "Normal" load levels, both static and dynamic, that should occur in a hydropower unit can be estimated from balancing standards, limitations regarding shape deviations in the generator, and experiences from the measured loads in a hydropower unit.

3.2 Balancing Grades. The ISO 1940-1 balancing standard gives recommendations for the maximum allowed unbalance. The allowed unbalance force is determined by the rotating mass, rotational speed, and balancing grade; see Eq. (6). These unbalance forces propagate to the radial bearing and, depending on the layout of the machine, the load distribution between the bearings differs. The most common bearing configurations for a hydropower unit consist of two or three radial bearings, where one of the bearings is a turbine guide bearing. For hydropower units equipped with three bearings, almost the entire load from the generator will be distributed between the two generator guide bearings and the load from the runner will be on the turbine guide bearing; see the layout in Fig. 9(*a*). When the machine is only equipped with two bearings, one is positioned close to the generator, above or below it, and the second bearing is positioned close to the runner, as described in the layouts in Figs. 9(*b*) and 9(*c*). In this configuration, almost all of the load from the generator will act on the

Table 2 Examples of included load spectra for components in a hydropower unit

Events	Number of load cycles	N_i
(1) Nominal operation start-stop cycles. The contribution from the maximum eccentricities are included.	$40 \times 360 \times 2$[a]	28,800
(2) Assumed number of short-circuits during lifetime multiplied by a factor of 6 (this includes connection out of synchronism)	10×6	60
(3) Assumed number of runaways during lifetime multiplied by a factor of 6	10×6	60
(4) Assumed number of load rejections during lifetime multiplied by a factor of 6	600×6	3600
(5) Mechanism for guide vane and rotor blade maneuvring, 10^7 load cycles from maximum closing force to maximum opening force		10^7
(6) Unbalance and generator eccentricities (occurring once per revolution)		$>10^7$

[a]40 years × 360 days × 2 start-stops /day.

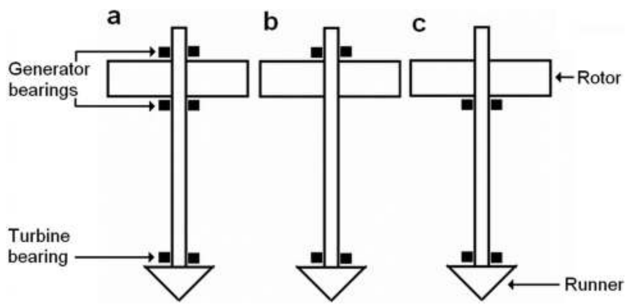

Fig. 9 Bearing layouts in hydropower unit units

generator guide bearing and almost all of the load from the runner will act on the turbine guide bearing.

The maximum allowed radial loads, according to ISO-1940, can be calculated from the chosen balancing quality grade, rotor mass, and rotational speed. The balancing quality grade determines the maximum allowed magnitude of the product of permissible residual specific unbalance e_{per} and the rotational speed Ω. According to ISO-1940, G6.3 is the recommended balancing quality grade for hydropower units, which implies that $e_{\text{per}} \cdot \Omega = 6.3$ mm/s. According to Eq. (6), the maximum allowed bearing loads at different rotor mass and rotor speeds for balancing quality grades G6.3 and G16 are presented in Table 3

$$f_u = \frac{m \cdot e_{\text{per}} \cdot \Omega^2}{1000} \tag{6}$$

where f_u is the radial force, m is the rotor mass, Ω is the rotor speed and e_{per} is the permissible residual specific unbalance.

3.3 Eccentricities in the Generator and Uneven Flow Properties in the Runner.

In addition to the fatigue criteria and balancing grades, the manufacturer/customer has recommendations for the maximum allowed shape deviations in the generator. Due to the unbalance magnetic pull (UMP) forces, an eccentric stator in relation to the rotor causes static load on the generator bearings. The rotor will also experience a cyclic load of the frequency $1 \times \Omega$, caused by the UMP due to the eccentric stator. A generator rotor center that is eccentric in relation to the shaft center will cause dynamic loads on the generator bearings and static loads on the rotor. Compared to generators in steam and gas power units, the UMP forces in a hydropower unit are high; normal UMP forces at 1 mm eccentricity are 200–400 kN. It is up to the customer to determine the maximum allowed eccentricities in a generator; common values are 3% maximum stator eccentricity and 1.5% rotor eccentricity.

In regard to the radial loads in the runner, it is more difficult to determine maximum allowed loads at normal operation. The balancing grade of the runner sets the maximum allowed dynamic loads due to mass unbalance. Static loads in the turbine are caused by uneven flow properties in the turbine. As stated in Ref. [14], "Measurements have been performed on Russian hydropower units, in the mid 20th century, in order to identify a "statistical" relationship between static radial loads and runner properties."

Table 3 Examples of bearing loads in symmetrically located rotors, such as in the generator bearings for the layout in Fig. 8(*a*)

Rotor mass (kg)	Rotor speed (RPM)	Bearing load G16 (kN)	Bearing load G6.3 (kN)
200,000	167	28	11
500,000	83	35	13
100,000	500	42	16

The following relationship for Francis turbines was identified from these measurements

$$f_r = \frac{8 \cdot \pi \cdot P}{300 \cdot \Omega \cdot D} \tag{7}$$

where f_r is the radial load from the runner, P is the rated power, Ω is the synchronous speed of the machine, and D is the runner diameter. This relationship can be used as a rule of thumb for expected static radial loads from the runner; however, the relationship still needs to be evaluated and it would be good if a more physical-related model existed. Studies regarding radial loads in pumps [15] have also showed that the radial loads are strongly dependent on the operating point of the pump.

Using the fatigue calculations, balancing grades, recommendations for shape deviations in the generator, and identified relationships between the runner properties and radial loads, it is possible to determine the radial loads for which the machine is dimensioned and which radial loads can be expected during normal operation.

4 Methodology for Vibration Monitoring

The methods for monitoring vibrations in a hydropower unit and the settings for allowed vibration levels should be based on the mechanical properties of the machine, the loads for which the machine is dimensioned, and the loads the vibrations cause.

4.1 General Methodology.

The methodologies presented in the following text propose a procedure to determine the alarm and trip levels based on the measured load levels and design properties of the machine.

4.1.1 Identify Properties of Mechanical Components in the Hydropower Unit. First, an analysis of the dimensioning prerequisites for the machine and the identification of critical components is needed. The fatigue limits for the critical components must be identified using the manufacturer design data or performing new FEA and fatigue calculations.

4.1.2 Relationship Between Vibration and Load Levels. Different methods can be used to determine bearing loads:

(a) Install load cells inside or behind the bearing. Measure the static and dynamic load.
(b) Calculate the bearing load from calculated bearing properties and measured shaft displacements.
(c) Calculate the bearing load from the stiffness properties of the bearing brackets and measured housing movement (it is only possible to determine dynamic loads using this method).

4.1.3 Identification of Maximum Recommended Load Levels During Normal Operation. Provided that a balancing standard (e.g., ISO 1940) is used, maximum load levels that fulfill the standard can be calculated. The loads caused by imbalance are dynamic loads.

For generators, the maximum eccentricity of the stator and rotor determines the maximum loads from the generator. The maximum allowed unbalance pull force is calculated by multiplying the air gap (normal air gap: 15–25 mm), eccentricity, and magnetic stiffness of the generator (normally $200\text{–}400 \cdot 10^6$ N/m). The loads are distributed between the upper and lower generator bearings if there are upper and lower generator guide bearings; otherwise, the single generator bearing will take all loads from the generator. The turbine guide bearing will take all loads caused by the runner.

4.1.4 Recommended Levels. The alarm and trip levels should be based on the operation mode, balancing grade, and fatigue limits. Under normal operating conditions, the radial load should not exceed the allowed dynamic load plus the maximum allowed static

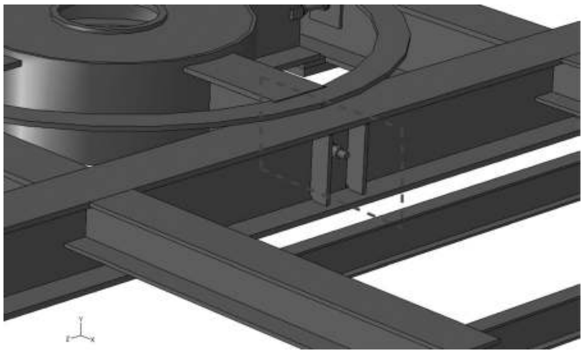

Fig. 10 Design of the upper bearing bracket for the hydropower unit presented in Fig. 1 and the critical bolt (dashed box)

load; the machine should send an alarm if these load levels are exceeded (normally at load levels between 75 and 125 kN). The trip level of the machine should be set with a large margin for the fatigue limit. Higher loads should be allowed during start up due to dynamic behavior in the runner. If only accelerometers are used, the static loads (from stator eccentricity, etc.) are not possible to determine. It is not recommended to only use accelerometers.

4.2 Results From the Methodology Applied to the Upper Guide Bearing in a 40 MW Hydropower Unit

4.2.1 Identify Properties of Mechanical Components in the Hydropower Unit. Finite element calculations identify the bolt connecting the inner and outer bearing bracket as the most critical component; see Fig. 10. The properties of the component are:

- stress area: $2 \times 817\ \text{mm}^2$ ($2 \times$ M36)
- fatigue cycles: 1 cycle/revolution $\Rightarrow N_i \sim 10^9$
- fatigue properties: the data sheet on the 5.6 (1550 steel) bolt gives the maximum allowed stress amplitude in the bolt as 35 MPa for $N_i = 10^9$ (when stress concentrations and safety margins are considered)
- maximum allowed stress and load: 35 MPa in the maximum allowed stress gives a maximum load of 570 kN

4.2.2 Relationship Between Vibration and Load Levels. The method for determining the bearing load from the calculated bearing properties and measured displacement was used. A bearing clearance check resulted in a bearing clearance of 0.4 mm. The results from the calculated bearing data as a function of the measured clearance and shaft displacement are presented in Fig. 11.

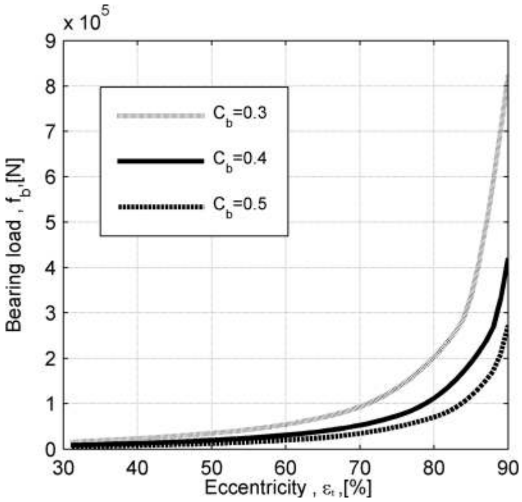

Fig. 11 Bearing properties at the present bearing clearance and at a ±25% change of bearing clearance

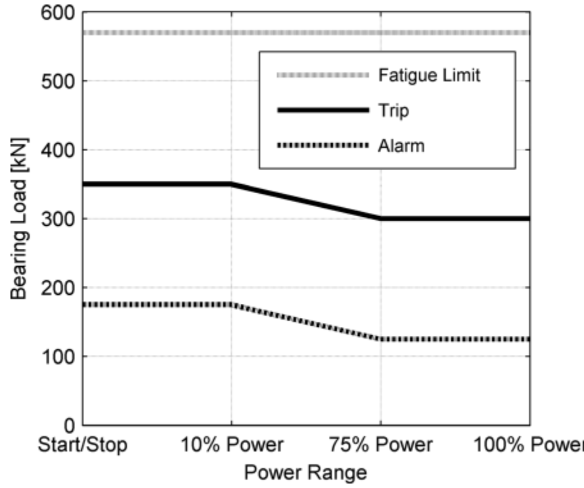

Fig. 12 Proposed levels for the alarm and trip

4.2.3 Identification of Maximum Recommended Load Levels During Normal Operation. The rotor mass of the unit is approximately 200 metric tons and the synchronous rotational speed is 167 rpm. Using the balancing grade of G6.3 results in a bearing load of 11 kN and 28 kN if the balancing grade of G16 is used.

The maximum recommended stator and rotor eccentricity, according to the owner of the hydropower unit, is 3% and 1.5%. The air gap is 20 mm and the linearized magnetic pull forces are 250×10^6 N/mm (valid for eccentricities less than 10% of the air gap).

The maximum static and dynamic pull forces distributed on each bearing are 75 kN and 38 kN.

4.2.4 Recommended Levels. The static loads are 75 kN and the dynamic loads are approximately 50 kN, depending on the balancing grade. During synchronous operation, the machine should send an alarm when loads exceed 125 kN (82% eccentricity at $C_b = 0.4$) and trip at 300 kN (88% eccentricity at $C_b = 0.4$). Figure 12 presents the recommended alarm and trip levels regarding radial loads in the upper generator guide bearing.

5 Discussion

A simple method with good accuracy for determining the radial load on the guide bearing in a hydropower unit does not currently exist. Load cells and strain gauges installed inside pivot pins offer a high level of accuracy for bearing load measurement, but these load measurement methods are elaborate and costly. Installing strain gauges on the supporting structure is easy, but the support structures in a hydropower unit generally consists of large steel beams. The large cross-section of the steel beams causes strain amplitudes during normal operation and the strain due to thermal expansion of the steel beams is often larger than the measured strains due to the radial loads. The method described in this paper, regarding determination of radial loads from calculated bearing data and measured shaft displacement, does not demand extensive installations and provides distinct load levels. The prerequisite for the method is that the user has access to the bearing properties of the machine or software to calculate the bearing parameters and knows the bearing clearance, the position of the bearing center, and the shafts position.

In regard to a monitoring system to protect the machine from harmful radial loads, the system should be chosen based on the importance and size of the machine. For large machines, i.e., larger than 100 MW, and perhaps with sliding stator feet or a "floating" rotor rim, avoiding damage is of very important. It is recommended that these machines be equipped with sensors for "direct" force determination, i.e., depending on the design of the hydropower unit, use the best suited of the methods presented in

Sec. 2. The machine's fatigue limits must also be identified and the alarm and trip levels must be set using the methodology proposed in the previous section. For old small machines, investing in a sophisticated vibration monitoring system is not justified; it is sufficient to install accelerometers or displacement sensors and to spend some time determining suitable alarm levels. For the mid size machines that often are large in number but have a rated power of less than 100 MW, a suitable monitoring method could be to use bthe earing properties and displacement measurements to determine the radial loads and use the methodology proposed in the previous section to identify the alarm and trip levels.

6 Conclusion

The methodology presented in this paper regarding condition monitoring is based on the mechanical properties of the critical components and the measured radial loads that act on these components. Using this methodology, it is possible to determine alarm and trip levels for the monitoring system based on the radial load levels in relation to the expected load levels during normal operation and the fatigue limits for the critical components. The paper also presents an alternative method for measuring the bearing load in hydropower units using calculated bearing parameters and shaft displacement measurements. This method is not expected to have the same high resolution as load measurement methods using the load cell, but the installation needed for the method is quick, easy, and does not require any modification of the components in the hydropower unit. It is also possible to determine both the static and dynamic radial loads using the method. Using the condition monitoring methodology and the method to determine the radial load presented in the paper gives better prerequisites to protect the machine from harmful radial loads and to avoid false alarms regarding vibrations.

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